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Vehicles Esp Control Performances on the Two-pitman Restriction Test System

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Abstract: Real vehicle test is a necessary segment in the procedure of vehicle ESP control system research and development, which can really shows the real effect of vehicle ESP control and is a strong mean and final test method in the research and development. ESP control performances exist some problems such as high dangerous, long period, high cost of the common ESP control performances real vehicle test. Project the two-pitman restriction test system, establish its dynamic system model, Take example for cheryA3 car, base on Matlab/Simulink, establish the two-pitman restriction test system, use the integrating braking and driving ESP control strategy, separately analyze and testify the ESP control performances in independent vehicle system and in the two-pitman restriction test system on three test conditions including neutral steering, under steering, over steering. The study results indicate that the ESP control performances of the vehicle system and the two-pitman restriction test system have remarkable consistency in three test conditions, laying a foundation of the ESP control test research.

Key words: Vehicle, ESP, test, simulation

INTRODUCTION

Vehicles ESP Control Performances test is an essential method, when a set of ESP system is developed and a real vehicle ESP control performances is detection. A lot of works have been done and there are many research results in this field in the world. The common research means and methods are simulation test (Wang *et al*., 2006; Guo *et al*., 2008; Chen *et al*., 2005; Liu and Hua, 2003a, b), test-bed research (Wang *et al*., 2006; Lin and Hua, 2003; Yu *et al*., 2007a, b; Piyabongkarn *et al*., 2009; Jo *et al*., 2008) and real vehicle test (Wang *et al*., 2007; Yang *et al*., 2007; Zhang *et al*., 2006, 2010). Computer simulation technology is a widely used method in primary research; test-bed technology is widely used to test some important components of dynamic performance in vehicle dynamic system in vehicle ESP control research; real vehicle test is a necessary segment in the procedure of vehicle ESP control system research and development, which can really shows the real effect of vehicle ESP control and is a strong mean and final test method in the research and development. Real vehicle test of vehicle ESP control need to be done in all kinds of complicated driving conditions, which have some shortages such as high dangerous, long test period, big invest, due to the limit of nature condition, safe protect condition, test ground and so on.

Now project the two-pitman restriction test system, establish its dynamic system model against the shortages of current vehicle ESP control test research. Take example for cheryA3 car, base on Matlab/Simulink, establish the two-pitman restriction test system, use the integrating braking and driving ESP control strategy, separately analyze and testify the ESP control performances in independent vehicle system and in the two-pitman restriction test system on three test conditions including neutral steering, under steering, over steering. The research results indicate that the ESP control performances of the two-pitman restriction test system and the independent vehicle system have great uniformity on the three test conditions, which proves that the two-pitman restriction vehicle system is safe and feasible in principle and overcomes the shortages of ESP control performances like high dangerous, long period, high cost of the common ESP control performances real vehicle test.

TWO-PITMAN RESTRICTION TEST SYSTEM

The two-pitman restriction test system, showed in Fig. 1, consists of two sets of adjustable pitman and one vehicle system. Each adjustable pitman includes casing, slide rod and spring. The slide rod moves front and back in the casing; there are two linear bearings

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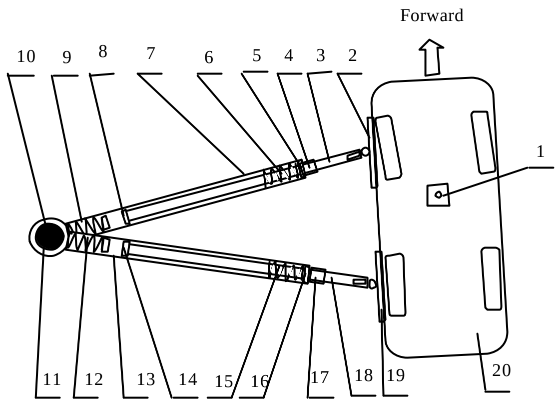

Fig. 1: Two-pitman restriction test system

between the slide rod and the casing to reduce the greasy friction. The end of the slide rod near the vehicle is mounted to vehicle clamp by use of spherical hinge and vehicle clamp is installed on vehicle. Two spherical hinges of adjustable pitman sets respectively lie in the front wheel axle and the rear wheel axle in longitudinal direction, as well as in incline axle of vehicle body in vertical direction to reduce the impact on vehicle incline; by pin roll, the ends of the front casing and the rear casing separately are fixed in the stationary shaft which is fixed in the ground; the end of front casing near the stationary shaft is attached an revolution speed transducer to measure its rotary speed; there is an angular displacement sensor between the front casing and the rear casing to measure the angle of two casing; the linear movement pick-up is built between the casing and slide rod to measure the expansion and contraction quantities.

When vehicle tends to instability with smaller turning radius in over steering, the slide rod is contracted to minimum in the casing. At the same time, there is a spring at the end of slide rod to limit it to go on being contracted, thus the severe vehicle flinging tail will be avoided. When the vehicle turning radius larges in under steering and causes vehicle to instability, the slide rod is expanded to maximum in the casing, the spring at other end of slide rod restricts the trend of vehicle under steering, which avoids vehicle losing control. The two-pitman restriction test system effectively prevents the dangerous happening in the test of vehicle ESP control and promotes the safety of test and research.

By means of the rotary speeds of the front casing and the rear casing, the angle between the front casing and the rear casing, the expansion and contraction quantities of the front casing and the rear casing, the angle of the front wheel turning and the sign of gyroscope, the motion

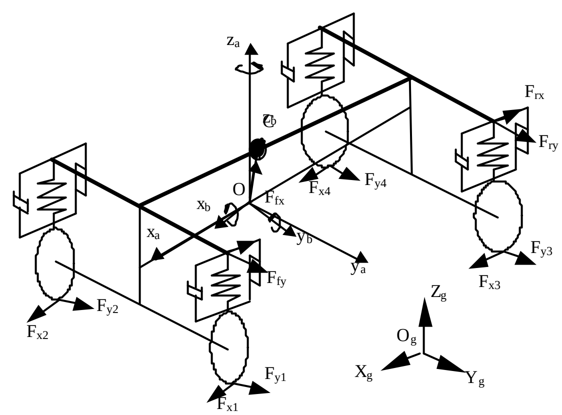

Fig. 2: Whole vehicle dynamic model

condition parameters of the vehicle would be measured, such as the longitudinal velocity of the vehicle mass, the lateral velocity of the vehicle mass, the longitudinal velocity of the each wheel center, the lateral velocity of the each wheel center, the slip angle of vehicle, etc. This prepared the ground for effectively testing the vehicle ESP control performance and researching the control strateg 1. gyroscope 2, 19. vehicle clamp 3. the front slide rod 4, 7. linear movement pick-up 5, 8, 14, 16. linear bearing 6, 9, 12, 15.spring 7. the front casing 10. stationary shaft 11.revolution speed transducer 13.the rear casing 18.the rear slide rod 20. vehicle

DYNAMIC MODEL

Dynamic model of the whole vehicle: The whole vehicle dynamic model is shown in Fig. 2, which is set up according to the needs of researching vehicles ESP

Table 1: Whole vehicle dynamic model parameters

Symbol	Meaning
u, v	The longitudinal and lateral velocity for origin of the vehicle coordinate system, m sec^{-1}
φ, θ, Ψ	The roll angle, pitch angle and yaw angle of the body, rad
Kφ, Kθ	The roll and the pitch stiffness of the body, N m rad^{-1}
Cφ, Cθ	The roll and pitch damping coefficient of the body, kg m^{-2} rad^{-2}
Fxi, Fyi, Fzi	The longitudinal force, lateral force and vertical force of the tire, N
Mx, My, Mz	The torque of the whole vehicle around three axises respectively, N m^{-1}
Iz	The rotational inertia of the whole vehicle around axis z, kg m^{-2}
Ixb, Iyb, Izb	The rotational inertia of the body around axis x, y, z, kg m^{-2}
Ixzb	The product of inertia for the body centroid to axis x, z, kg m^{-2}
Izf, Izr	The non-sprung mass yaw rotational inertia of the front and rear car bridges, kg m^{-2}
hb	The height between the body centroid and the origin of the body coordinate system, m
hg	The height between the centroid of the whole vehicle and the ground, m
a, b	The longitudinal distance from the front and rear axle to the body centroid, m
Bf, Br	The front and rear wheelspan, m
ma, mb, mf, mr	The mass of the whole vehicle and the body, the non-sprung mass of the front and rear car bridge, kg
g	The acceleration of gravity, m sec^{-2}
δ1, δ2	The steering angle of the inside and outside front wheel, (o)
α1, α2, α3, α4	The slip angle of the inside and outside front wheel and the slip angle of the inside and outside rear wheel, (o)
Cα1, Cα2, Cα3, Cα4	The cornering stiffness of the inside and outside front wheel and the cornering stiffness of the inside and outside rear wheel, N·rad-1

control performances. Og-XgYgZg is the ground coordinate system, which is the reference base G; O-xayaza is the whole vehicle coordinate system, which is the reference base A; O-xbybzb is the body coordinate system, which is the reference base B. The origin of the body and the whole vehicle coordinate system is the body roll center point O within the cross-section through the vehicle centroid. The level line through point O and in the longitudinal symmetry plane is axis xa, which the forward direction is positive; the level line through point O and in the vehicle cross-section is axis ya, which the vehicle left side is positive; the line through point O and vertically upwards is axis za. The axis direction of the body coordinate system is the same as the whole vehicle coordinate system when the vehicle is in stationary. The centroid of the vehicle mass above the vehicle suspension sprung is point C, the origin of ground coordinate system is the ground projection point of point C when the vehicle is in stationary, the axis direction is the same as the whole vehicle coordinate system (Liu and Hua, 2003a, b; Wang *et al*., 2006).

The 10-DOF vehicle dynamic model is shown in Fig. 2. The speed of the whole vehicle along the longitudinal axis is u; the speed along the lateral axis is v; the rotation velocity of the whole vehicle around the axis z_a is ω_z. The body roll angle around the longitudinal axis x_b relative to the whole vehicle coordinate system is φ and the pitch angle around the lateral axis relative to the whole vehicle coordinate system is θ. The rotation velocities of the four wheels around the their axis are ω_i (i = 1, 2, 3, 4); the steering angle of inside front wheel is ä1. The model is set up for the foundation of research for the vehicle stability control performances, fully considering of the vehicle longitudinal movement, the yaw movement, the lateral movement, the roll movement and pitch movement of the body.

In Fig. 2, F_{fx} is the vehicle longitudinal force of the front pull rob on the vehicle, F_{fy} is the vehicle lateral force of the front pull rob on the vehicle, F_{rx} is the vehicle longitudinal force of the rear pull rob on the vehicle, F_{ry} is the vehicle lateral force of the rear pull rob on the vehicle, which are lying on the high of the centroid point C. The whole vehicle dynamic model parameters are shown in Table 1.

In the whole vehicle dynamic model, the external forces of whole vehicle are simplified as the wheel longitudinal, lateral and vertical forces F_{xi}, F_{yi}, F_{zi} (i = 1, 2, 3, 4), which are the ground given to the wheels, the front pull rod's forces F_{fx}, F_{fy} and the rear pull rod's forces F_{rx}, F_{ry} on the vehicle body. Thus we can get the vehicle resultant forces along the longitudinal axis x_a and lateral axis y_a, which are assumed as F_x, F_y, the resultant torques around the longitudinal axis x_a and lateral axis y_a and the vertical axis z_a are assumed as M_x, M_y, M_z, respectively. The whole vehicle dynamic equation according to Lagrange equations can get as follows:

$$
\begin{aligned}
& m(\dot{u} - v\dot{\psi}) + m_b h_b \ddot{\theta} = F_x \\
& m(\dot{v} + u\dot{\psi}) + (am_f - bm_r)\ddot{\psi} + m_b h_b \ddot{\phi} = F_y \\
& K_\phi \phi + C_\phi \dot{\phi} + m_b(g\phi + \dot{v} + 2h_b\ddot{\phi} + h_b\ddot{\theta} + u\dot{\psi})h_b + I_{xb}\ddot{\phi} = M_x \\
& K_\theta \theta + C_\theta \dot{\theta} + m_b(\dot{u} + 2h_b\ddot{\theta} + h_b\ddot{\phi} - v\dot{\psi} + g\theta)h_b + I_{yb}\ddot{\theta} = M_y \\
& m_b(\dot{v}\theta + v\dot{\theta} - \dot{u}\phi - u\dot{\phi})h_b + m_f a(\dot{v} + a\ddot{\psi}) - m_r b(\dot{v} - b\ddot{\psi}) + I_z\ddot{\psi} = M_z
\end{aligned}
$$

where, the parameters are shown in Table 1, the equations are still the non-linear equation although it ignores the higher order terms.

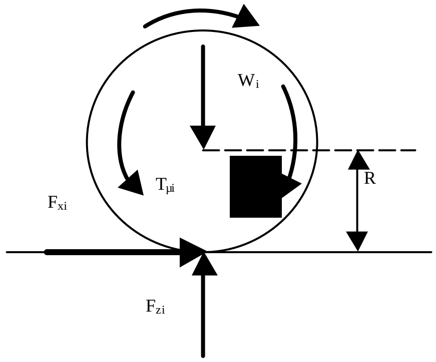

Fig. 3: Dynamic analysis for wheel

The wheel longitudinal and lateral forces F_{xi}, F_{yi} (i = 1, 2, 3, 4) given by the ground can be obtained by the longitudinal slip and lateral slip uniform ground-tire model (Lin and Hua, 2003). The wheel dynamic vertical force $F_z = [F_{z1}\ F_{z2}\ F_{z3}\ F_{z4}]T$ is as follows:

$$F_z = \begin{bmatrix} \frac{m_a gb}{2L} + \frac{F_x h}{2L} - \frac{F_y h}{2B} \\ \frac{m_a gb}{2L} + \frac{F_x h}{2L} + \frac{F_y h}{2B} \\ \frac{m_a ga}{2L} - \frac{F_x h}{2L} - \frac{F_y h}{2B} \\ \frac{m_a ga}{2L} - \frac{F_x h}{2L} + \frac{F_y h}{2B} \end{bmatrix}$$

where, L is the wheelbase, m; B is the average value of the front and rear wheelspan, m; Other parameters are same as before.

The dynamic analysis (Yu *et al*., 2007a, b) of the vehicle tire is shown in Fig. 3. The torque balance equation of the wheel is:

$$I_w \dot{\omega}_i = T_{qi} - F_{xi}R - T_{\mu i}$$

where, I_w is the rotational inertia of the wheel, kg m^{-2}; ω_i (i = 1, 2, 3, 4) is the rotation velocity of the wheel, rad sec^{-1}; R is the wheel rolling radius, m; T_{qi} (i = 1, 2, 3, 4) is the driving torque of the wheel, N.m; T_{ii} is the braking torque of wheel, N m; Other parameters are same as before.

The equivalent dynamic model of the steering system turning around knuckle pin is shown in Fig. 4.

The steering system dynamic equation is:

$$(I_h + \frac{I_s}{i^2})\frac{d^2\theta_h}{dt^2} + (c_h + \frac{c_s}{i^2})\frac{d\theta_h}{dt} = T_h - \frac{T_s}{i}$$

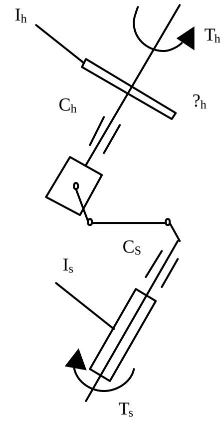

Fig. 4: Equivalent dynamic model of the steering system turning around knuckle pin

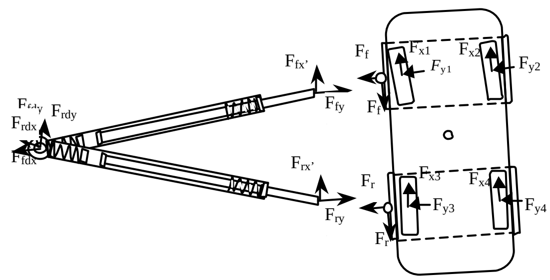

Fig. 5: Two-pitman restriction test system dynamic model

where, I_h is the rotational inertia of the steering wheel, kg m^{-2}; θ_h is the steering angle of the steering wheel, rad; T_h is the driving torque of the steering wheel, N m; I_s is the rotational inertia of the front wheels and the steering mechanism simplified to the knuckle pin, kg m^{-2}; Ts is the steering resistance torque of the front wheels, that is the aligning torque, N.m; C_h is the equivalent friction coefficient between the steering column and the supports, C_s is the equivalent friction coefficient between the front wheels and the steering mechanism; i is the transmission ratio from the steering column to the front wheels.

Dynamic model of the test system: The dynamic model of the two-pitman restriction test system is shown in Fig. 5. F_{fdx}, F_{fdy} are the forces applied to the front casing by the stationary shaft, N; F_{rdx}, F_{rdy} are the forces applied to the rear casing by the stationary shaft, N; F_{fx}, F_{fy} are the longitudinal force and lateral force applied to vehicle body by the front slide rod, N; Likewise, the forces caused by the rear casing and the rear slide rod are gotten. According to the expansion and contraction quantities of the front slide rod and the rear slide rod, the angular displacement between the front casing and the rear casing, the relationship between two slide rods and vehicle can be decided. F_{xi}, F_{yi} (i = 1, 2, 3, 4) are the longitudinal force and lateral force of four wheels, N.

The dynamic model of the two-pitman restriction test system is listed:

$$I_f \dot{\omega}_f = M_{Ffx} + M_{Ffy} + M_{ff}$$
$$I_r \dot{\omega}_r = M_{Frx} + M_{Fry} + M_{fr}$$

I_f, I_r are the rotary inertia of the front and rear slide rods separately rotating the stationary shaft, kg.m2; ω_f, ω_r are the rotary angle speed of the front and rear slide rods separately rotating the stationary shaft and can be found out by means of the angular displacement between the front casing and the rear casing, the rotary speed of the front casing end, rad/s; M_{Ffx}, M_{Ffv}, M_{Frx}, M_{Frv} are the torques of F_{fx}, F_{fy}, F_{rx}, F_{ry} applied to the stationary shaft, Nm, meanwhile, the arm of force relative to the stationary shaft can be gotten by use of the relationship between the front and rear slide rods and vehicle; M_{ff}, M_{fr} are torques caused by friction at joints in the two sets of adjustable pitman, Nm and can be measured by test.

Through introducing the two sets of adjustable pitman to the dynamic model of independent vehicle system and analyzing the forces (F_{fx}, F_{fy}, F_{rx}, F_{ry}) of the front and rear slide rods acting on the vehicle, the dynamic model of two-pitman restriction test system is established.

ANALYZE THE PERFORMANCE OF TEST SYSTEM

For verify the stability control performance of the two-pitman restriction test system, based on Matlab/Simulink, using the brake/drive integrated ESP control principle, adopting vehicle parameters of the CheryA3 car, a simulation test system of the two-pitman restriction test system is built to separately test, analyze and prove the ESP control performances in different test conditions for different systems-independent vehicle system and the two-pitman restriction test system.

In the condition under steering or over steering, the research for the stability control test of the independent vehicle system respectively is engaged with ESP, or without ESP. Through adjusting the ESP control strategy, the independent vehicle system gets the good neutral steer. Based on the same conditions and ESP control strategy, the research for the stability control test of the two-pitman restriction test system is also engaged. The stability control effects are judged by the motion path of the center of vehicle mass and vehicle side slip angle. Figure 6, 8 are the motion path curves of the center of vehicle mass, where the x-coordinate is the motion path horizontal movement of the center of vehicle mass in the earth coordinate system and the y-coordinate is its motion path vertical movement, m; Fig. 7, 9 are the side slip angle curves of the vehicle center mass, where the x-coordinate is the motion path horizontal movement of the center of vehicle mass, m and the y-coordinate is the side slip angle of the vehicle center mass, deg. In Fig. 6-9, the continuous thick lines are the performance curves of

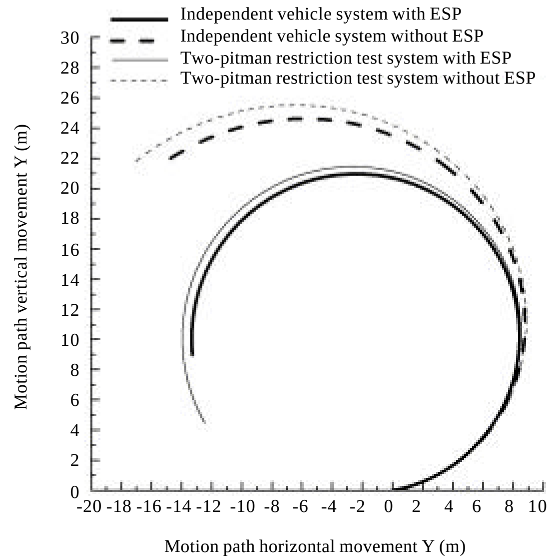

Fig. 6: Vehicle center mass paths under steering

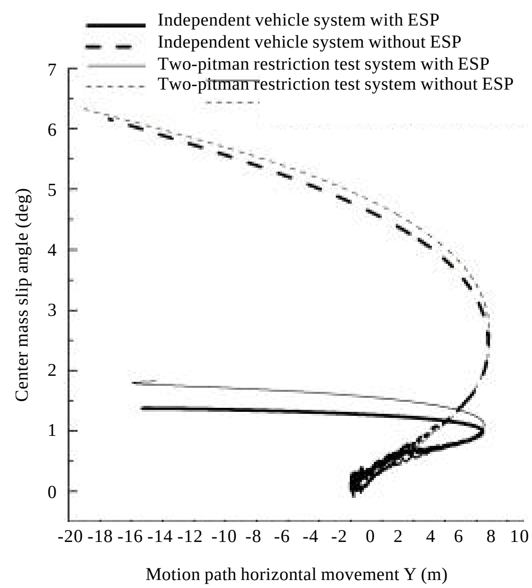

Fig. 7: Vehicle center mass side slip angle under steering

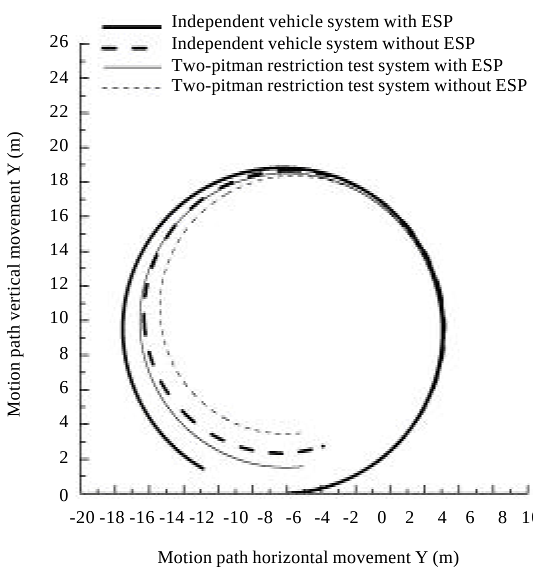

Fig. 8: Vehicle center mass paths over steering

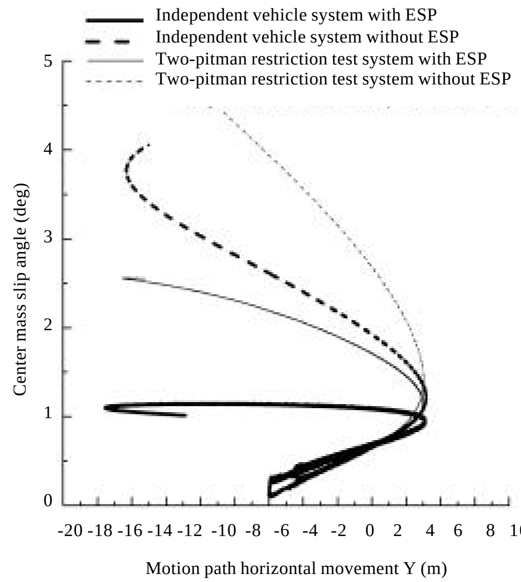

Fig. 9: Vehicle center mass side slip angle over steering

the dependent vehicle system with ESP, regarded as the neutral steering curve; the thick dotted lines are the performance curves of the dependent vehicle system without ESP; the continuous thin lines are the performance curves of the two-pitman restriction test system with ESP; the thin dotted lines are the performance curves of the two-pitman restriction test system without ESP.

Analysis of performance under steer: The simulation conditions: The front wheel angle of vehicle is 15°; the initial speed is v_0 = 5 km h^{-1}, the final speed is v_t = 60 km h^{-1}; the test road is bisectional road, where the adhesion coefficient between tire and road in the inner side is 0.8 and the adhesion coefficient between tire and road in the outer side is 0.2; the initial lengths of the front and rear slide rods in the two-pitman restriction test system are $L_f = L_r$ = 10m; through designing the strength and pressure bar stability of the two-pitman restriction test system, the rotational inertias of the front and rear slide rods relative to stationary shaft are If Ir = 1209.2 kg m^{-2}.

Figure 6 is the path of the vehicle center mass under steering; Fig. 7 is the vehicle side slip angle under steering. When completing test, the maximum curvature radius of the vehicle center mass path in the two-pitman restriction test system without ESP is 27.459 m and the maximum side slip angle in the independent vehicle system without ESP is 6.1586°, which all are in the reasonable range. From the figures, based on the same stimulation conditions, the independent vehicle system with ESP is neutral steer, the independent vehicle without ESP is under steer; the two-pitman restriction test system with ESP is reasonable under steer, the two-pitman restriction test system without ESP is larger under steer. This proves that the stability control performance of the two-pitman restriction test system is significant consistency with that of the independent vehicle system under steering. At the same time, the two-pitman restriction test system tends to properly increase understeer. That is the properly larger system overshoot to help the control system research. Meanwhile, the unreasonable bigger understeer would be efficiently avoided to ensure test safety resulting from the limitation of the front and rear slide rods.

Analysis of performance over steert: The simulation conditions: The front wheel angle of vehicle is 15°; The initial speed is v_0 = 5 km h^{-1}, the final speed is v_t = 60 km h^{-1}; the test road is bisectional road, where the adhesion coefficient between tire and road in the inner side is 0.1 and the adhesion coefficient between tire and road in the outer side is 0.9; other simulation conditions are the same with the mentioned.

Figure 8 is the path of the vehicle center mass over steering; Fig. 9 is the vehicle side slip angle over steering. When completing test, the maximum curvature radius of the vehicle center mass path in the two-pitman restriction test system with ESP is 11.251m that is reasonable and the maximum side slip angle in the two-pitman restriction test system without ESP is 4.4581° that tends to increase. From the figures, based on the same stimulation conditions, the stability control performance of the two-pitman restriction test system is significant consistency with that of the independent vehicle system over steering. At the same time, the two-pitman restriction test system tends to properly increase oversteer. That is the properly larger system overshoot to help the control system research also. Meanwhile, the unreasonable bigger oversteer would be efficiently avoided to ensure test safety resulting from the limitation of the front and rear slide rods.

CONCLUSION

Design a two-pitman restriction test system to set up a safe and efficient test system for vehicle motion stability control. Taking example for cheryA3 car, based on Matlab/Simulink, the dynamic s I m u lation system of the two-pitman restriction test system is establish with using the integrating braking and driving ESP control strategy, then separately analyze and testify the ESP control performances in independent vehicle system and in the two-pitman restriction test system on three test conditions including neutral steering, under steering, over steering. The research conclusions are as follows:

- The stability control performance of the two-pitman restriction test system is significant consistency with that of the independent vehicle system under steering. The two-pitman restriction test system tends to properly increase understeer, which results to properly increase the overshoot in the system
- Also, the stability control performance of the two-pitman restriction test system is significant consistency with that of the independent vehicle system over steering. The two-pitman restriction test system tends to properly increase overrsteer, which results to properly increase the overshoot in the system.

The research results show that the projected two-pitman restriction test system and its dynamic model for researching vehicle motion stability control test is feasible in theory and high safe against the shortages of the common methods for ESP control test, such as big dangerous, long period, high cost, etc.

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